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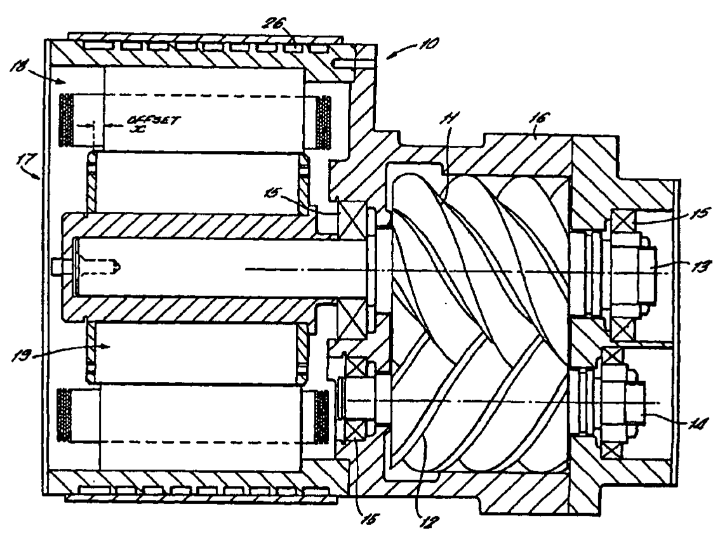
(54) Title: SCREW COMPRESSOR WITH SWITCHED RELUCTANCE MOTOR



(57) **Abstract:** The invention relates to improvements in compressors and in particular to arrangements for driving screw compressors by reluctance drive motors. A screw compressor assembly is provided for compressed gas comprising a screw compressor having a casing (16) and at least one pair of intermeshing helically formed screw rotors (11,12) each of which is mounted on a rotor shaft rotatably mounted within the casing by means of bearings (15). Said assembly further comprises a switched reluctance motor comprising a stator (18) attached to the casing and a rotor (19) mounted on one of the motor shafts such that one of the screw rotors is driven directly by the motor. The motor rotor is displaced relative to the stator by a pre-determined axial distance x) such that a resultant magnetic force is provided in an axial direction.

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SCREW COMPRESSOR WITH SWITCHED RELUCTANCE MOTOR

The invention relates to improvements in compressors and in particular to arrangements for driving screw compressors by switched reluctance drive motors.

Screw compressors are used in a variety of different applications for delivering compressed air, or other gasses, at a required pressure and volume. The speed at which the one or more stages of the compressor is driven is determined by the delivery requirements. Induction motors are currently used to drive the majority of industrial screw compressors. However, the design of such motors has a major effect on its behaviour and performance and seldom has it been possible to match the compressor speed with the speeds available from standard induction motors. To overcome this problem a belt drive system or gearbox is used. Then, by changing the drive ratio, a number of discrete outputs may be obtained for running a compressor at different speeds.

Where gears and belts are used to drive the compressor substantial extra loads are added to the compressor bearings due to the tension forces in the belts or the drive loads from the gears. There is also a reduction in efficiency as the belts or gears incur a drive loss.

More recently the use of inverter drives with induction motors has meant that standard motors may be run at infinitely variable speeds within the capabilities of the motors. Due to the use of induction motors this still means that there is an upper limit on the speed determined by the design of the motor. For example two pole motors run normally

at around 3600 rpm on a 60Hz supply with the use of an inverter they may be run to perhaps 4500 rpm, depending on the motor design. In small screw compressors speeds in excess of this are commonly required.

In most cases the induction motor is a self-contained unit coupled to the compressor through a flexible coupling.

Also, generally speaking induction motors are air-cooled which, when used with a rotary compressor, gives rise to fan losses and can lead to possible damage to the motor due to air borne contaminants. The noise generated by the fan is a significant contributor to the noise of the motor especially at high speeds.

EP-A-1041289 recognises the need to dispense with a gear assembly in a compressor assembly and to provide a direct link between the compressor and motor. It therefore describes a rotary compressor driven by a switched reluctance motor, wherein the impeller of the compressor is mounted on a drive shaft assembly, on which the rotor of the switched reluctance motor is also mounted.

However EP-A-1041289 is specifically directed towards a two stage centrifugal or axial flow compressor as opposed to a screw compressor. Centrifugal compressors operate in a very different speed range to screw compressors, generally in excess of 50,000 rpm, whereas most screw compressors operate at below 25,000 rpm.

It is therefore an object of the present invention to provide a screw compressor assembly which

overcomes these disadvantages by the use of a switched reluctance drive motor, where the rotor of the motor is mounted directly on an extension of the compressor drive shaft. Dispensing with the flexible coupling between the motor and compressor also leads to a saving in cost and enables a more compact machine to be produced.

The invention therefore provides a screw compressor assembly for providing compressed gas comprising a screw compressor having a casing and at least one pair of intermeshing helically formed screw rotors each of which is mounted on a rotor shaft rotatably mounted within the casing by means of bearings, said assembly further comprising a switched reluctance motor comprising a stator attached to the casing and a rotor mounted on one of the rotor shafts such that one of the screw rotors is driven directly by the motor, wherein the motor rotor is displaced relative to the stator by a predetermined axial distance such that a resultant magnetic force is provided in an axial direction.

The design of the motor can be matched to the characteristics of the screw compressor in terms of torque and speed. The switched reluctance motor may also be designed to run at almost any speed and its speed is infinitely variable within its designed speed range. An additional benefit is that the axial magnetic forces available from the rotor of the motor are used to affect the axial bearing loads in the compressor.

Where the application is in liquid injected compressors the cooling liquid for the compressor may be circulated around the stator of the motor to provide cooling for the motor. This has the benefit

over an air-cooled motor of being able to seal the motor thus reducing noise (especially important in high-speed drives) and improving reliability.

A preferred embodiment of the invention will now be described, by way of example only, with reference to the accompanying drawings, in which:-

Fig. 1 is a cross-sectional front elevation of a section of a rotary screw compressor according to the invention showing the mounting of the switched reluctance drive motor with the motor rotor displaced away from the compressor;

Fig. 2 is a schematic of the screw compressor of the present invention; and

Fig. 3 is similar to Fig. 1, but shows the motor rotor displaced in the opposite direction.

The embodiment of the invention illustrated in Figs. 1 and 2 is a single stage liquid-injected screw compressor assembly, although most of the features may be applied to an oil-free screw compressor or other types of single or multi-stage screw compressor. Fig. 1 shows a section of a single stage liquid-injected screw compressor (10) consisting of a pair of contra-rotating, helically cut fluted rotors (11, 12) mounted respectively on rotor shafts (13, 14) which are supported at each end in rolling bearings (15) in a rigid compressor casing (16). One of the rotor shafts (13), on which a "driven" rotor (11) is mounted is extended with respect to the other rotor shaft (14).

The drive motor (17) used to drive the compressor (10) is a switched reluctance motor. Switched reluctance motors suitable for use in the present

invention are already known in the art, such as described in EP-A-0702448. In this type of motor both the stator (18) and the rotor (19) have projecting poles. The rotor (19) has no permanent magnets or windings. Thus, when one of the phases of the stator (18) is energised, the closest set of poles of the rotor (19) are pulled into alignment. As each consecutive phase is energised, the rotor (19) is caused to rotate. Switched reluctance machines with different numbers of phases are well known in the art.

In the present invention, the rotor (19) of the switched reluctance motor (17) is attached to the extended rotor shaft (13) using a locking system, such as a taper or a parallel shaft with a key, to ensure concentricity and transmission of torque. The rotor (19) has no bearings and is overhung and therefore entirely supported by the compressor bearings (15). The stator (18) is attached to the compressor casing (16) so that it is concentric to the motor rotor (19).

Referring to Fig. 2, in operation a magnetic field is generated in the motor stator (18) which causes the motor rotor (19) to turn which rotates the driven compressor rotor (11). The motor (17) is connected to a power converter (20), in which a three phase electrical supply is rectified to produce dc voltage, which is then switched to the stator windings in a controlled manner to generate the necessary torque to produce the required rotational speed. The speed may be set to a fixed value manually or varied to maintain a specific value, calculated by a controller (30) to maintain a parameter such as compressor pressure within certain pre-set limits.

The rotation of the driven compressor rotor (11) transfers drive to the second rotor (12). This action

induces air into the compressor (10) through a filter (32) via an inlet port at an inlet end of the compressor (10). As the rotors (11, 12) continue to rotate the reduction in the volume between them and the casing (16) causes the trapped volume of air to be compressed. The compressor (10) is preferably driven at a speed lying in the range of 500 to 25,000 rpm and more preferably in the range of 1000 to 10,000 rpm. The compressed air is expelled via a delivery port at the delivery end of the compressor (10).

In this embodiment of the invention liquid is injected into the compressor (10) during compression to cool, seal and lubricate the compressor (10). The compressed air/liquid mixture is passed to a separator vessel (21) where the liquid is separated and the air leaves the compressor (10) at a compressor outlet (22) usually through a non-return valve (23) and aftercooler (24). A small amount of liquid which has coalesced in the separator (21) is returned to the compressor (10) via a scavenge line (33) and strainer (34).

The rest of the coolant liquid is passed, due to differential pressure in the system, from the separator vessel (21) through a radiator (25) and is then used to cool the stator (18) of the motor by passing through a cooling jacket (26), which is in close proximity to the stator windings. From the motor cooling jacket (26) the coolant is passed through a filter (27) and then back to the compressor (10) where the coolant is injected into the compressed air and is used to lubricate the compressor bearings (15).

In another embodiment of the invention, the motor may be air-cooled using a fan that would pass ambient

air over or through the motor windings.

To control the speed of the motor, pressure is sensed at the compressor delivery point by a pressure transmitter (31). This signal is passed to the controller (30) which in turn operates the power converter (20) to increase or reduce the speed of the motor until the desired pressure is reached.

Within a rotary screw compressor (10) the bearings (15) that support the rotors (11, 12) are designed to absorb radial and axial loads. The radial and axial loads are produced from a number of sources including gas loads, drive force resultants and the weight of the rotating parts. In a screw compressor (10) the dominant load is often an axial force on the rotors (11, 12) acting from the delivery end towards the inlet end of the rotors (11, 12). Traditionally the load on the axial load-carrying bearings (15) has been off-set by a number of means such as using the axial component of the thrust of a helical drive gear or by using a balance piston arrangement fed by either pressurised air or lubricant. However with the drive arrangement used in the present invention the axial loads are offset by arranging the position of the motor rotor (19) within the stator (18) so that there is a resultant magnetic force acting in the desired direction. This force can be generated where the motor rotor (19) is designed to run offset from the centre of the magnetic field generator in the stator windings. This force can be produced to act in either direction depending on the axial position of the rotor (19) relative to the stator (18). In the preferred embodiment of the invention this force acts towards the delivery end of the driven rotor (11) to relieve the effects of the axial gas thrust on the delivery end axially loaded bearing of the driven rotor (11).

Where the axial force exerted by the stator flux on the motor rotor (19) is used to reduce the axial load on the driven compressor rotor bearings (15) whilst running on load, as shown in Fig. 1, the motor rotor (19) is displaced axially away from the compressor rotors (11, 12) by a small distance. This distance would be determined by experiment for a given motor design. Motors where the rotor (19) is designed to be central in the stator (18) to have neutral axial forces are well known in the art. The amount of unbalanced magnetic pull (UMP) would be designed to equate to about 50% of the calculated axial load on the compressor bearings (15) at the highest load condition. Thus when the motor (17) is energised, the gas loads caused by compression will act towards the compressor inlet end, that is towards the motor (17) and the axial loads generated by the flux in the motor (17) will be opposite, that is towards the compressor rotors (11, 12). It is known in the art that the axial forces generated in the motor (17) increase as the applied torque increases. Similarly in the compressor rotors (11, 12) the axial loads on the bearings (15) will increase with increasing pressure and speed which corresponds with torque. Thus the load on the compressor axial bearings (15) caused by the gas loads and drive load components will be offset by the opposite acting magnetic loads from the motor rotor (19).

An enhancement of this embodiment of the invention is where the axial forces on the motor rotor (19), which are known in the art to be cyclic, are produced so as to be in phase with the cyclic axial loads from the compressor rotors (11, 12). In a screw compressor (10) the maximum axial load is known to take place at the point before the trapped volume of gas is released into the delivery port. The number of

times this takes place in a revolution is dependent on the number of lobes on the male rotor, i.e. for a four-lobed male rotor there will be four peaks per revolution. Similarly in the motor (17) the peak axial force takes place when the maximum flux is acting upon the motor rotor (19), so for a rotor (19) with four poles on it this will take place four times per revolution. Thus by arranging the points of peak load in the compressor rotors (11, 12) to coincide with the points of peak axial load acting in the opposite direction in the motor rotor (19), the forces are in phase. Thus the maximum benefit may be obtained in reducing the loads on the axial load carrying bearings.

The axial force exerted by the stator (18) flux on the motor rotor (19) can also be used to increase the load on the axial bearings (15) in the compressor (10) on start-up to ensure that the rotor (19) does not contact the end face of the compressor casing bore. This is shown in Fig. 3. In compressors where a single taper roller bearing (15) is used to control both radial and axial forces on the driven rotor (11) the weight of the rotor (11) can tend to allow it to drop when stationary due to the slope of the bearing race. This can then allow the end of the rotor (11) to contact the casing. Under conditions of marginal lubrication this can result in the rotor (11) contacting the casing and causing wear during starting. When running the lubricant is present and the dynamic loads will return the rotor to the appropriate running position in the bearing.

In this embodiment of the invention the motor rotor (19) is designed to be offset in the stator (18) by an amount corresponding to the movement of the rotor in the bearing (typically <0.2mm). Thus on

starting, the flux in the winding draws the motor rotor (19) with the compressor rotor (11) attached towards the inlet end of the compressor (10) thus moving the compressor rotor (11) away from the delivery end face of the casing bore. As the motor rotor (19) returns to its neutral axial position the axial forces diminishes and the compressor (10) runs normally.

CLAIMS:

1. A screw compressor assembly for providing compressed gas comprising a screw compressor having a casing and at least one pair of intermeshing helically formed screw rotors each of which is mounted on a rotor shaft rotatably mounted within the casing by means of bearings, said assembly further comprising a switched reluctance motor comprising a stator attached to the casing and a rotor mounted on one of the rotor shafts such that one of the screw rotors is driven directly by the motor, wherein the motor rotor is displaced relative to the stator by a predetermined axial distance such that a resultant magnetic force is provided in an axial direction.

2. A screw compressor assembly as claimed in claim 1 in which the motor rotor is positioned relative to the stator to direct the resultant magnetic force towards a delivery end of the compressor.

3. A screw compressor assembly as claimed in claim 1 in which the motor rotor is positioned relative to the stator to direct the resultant magnetic force to an inlet end of the compressor.

4. A screw compressor assembly as claimed in any one of the preceding claims in which the resultant magnetic force is directed to change a load on the screw rotor bearings of the driven screw rotor.

5. A screw compressor assembly as claimed in any one of the preceding claims in which cyclic axial forces on the motor rotor are arranged so as to be in phase with cyclic axial loads produced in the screw rotors.

6. A screw compressor assembly as claimed in any one

of the preceding claims further comprising a cooling system, comprising means for injecting a liquid coolant into the compressor, during operation, to cool, seal and lubricate the compressor and bearing means, wherein the coolant is also used to cool the switched reluctance motor.

7. A screw compressor assembly as claimed in claim 6 in which the cooling system comprises a cooling jacket surrounding the switched reluctance motor for receiving a flow of coolant liquid.

8. A screw compressor assembly as claimed in any one of the preceding claims comprising a controller for controlling the speed of the compressor, such that the volume and/or pressure of compressed gas delivered by the screw compressor assembly lies between predetermined limits.

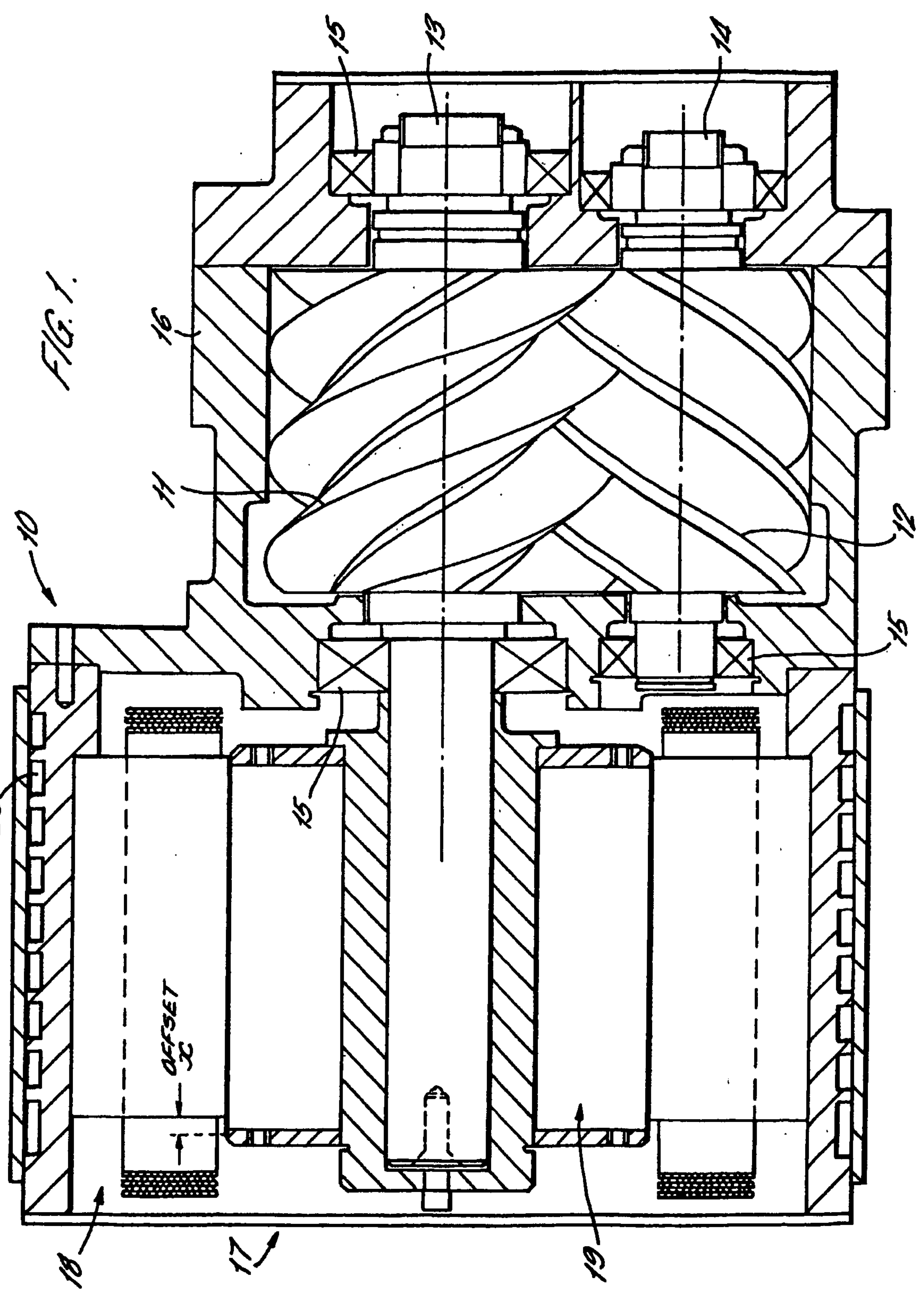
9. A screw compressor assembly as claimed in any one of the preceding claims in which the rotor of the motor is overhung on the bearings.

10. A screw compressor assembly as claimed in any one of the preceding claims in which the speed at which the compressor is driven lies in the range of 500 to 25,000 rpm.

11. A screw compressor assembly as claimed in claim 10 in which the speed at which the compressor is driven lies in the range of 1000 to 10,000 rpm.

12. A screw compressor assembly substantially as hereinbefore described with reference to and as shown in the accompanying drawings.

FIG. 1.



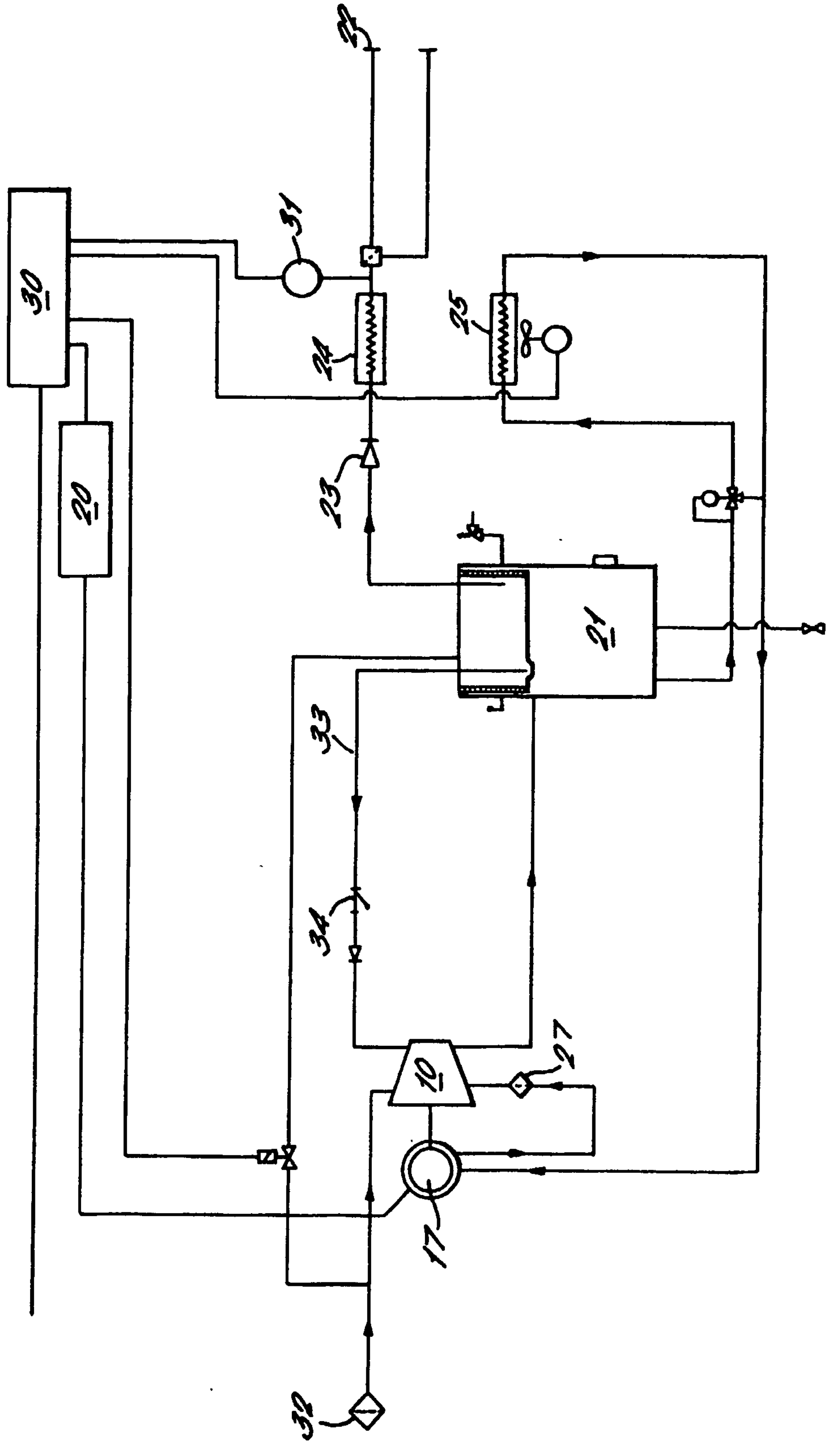


FIG. 2.

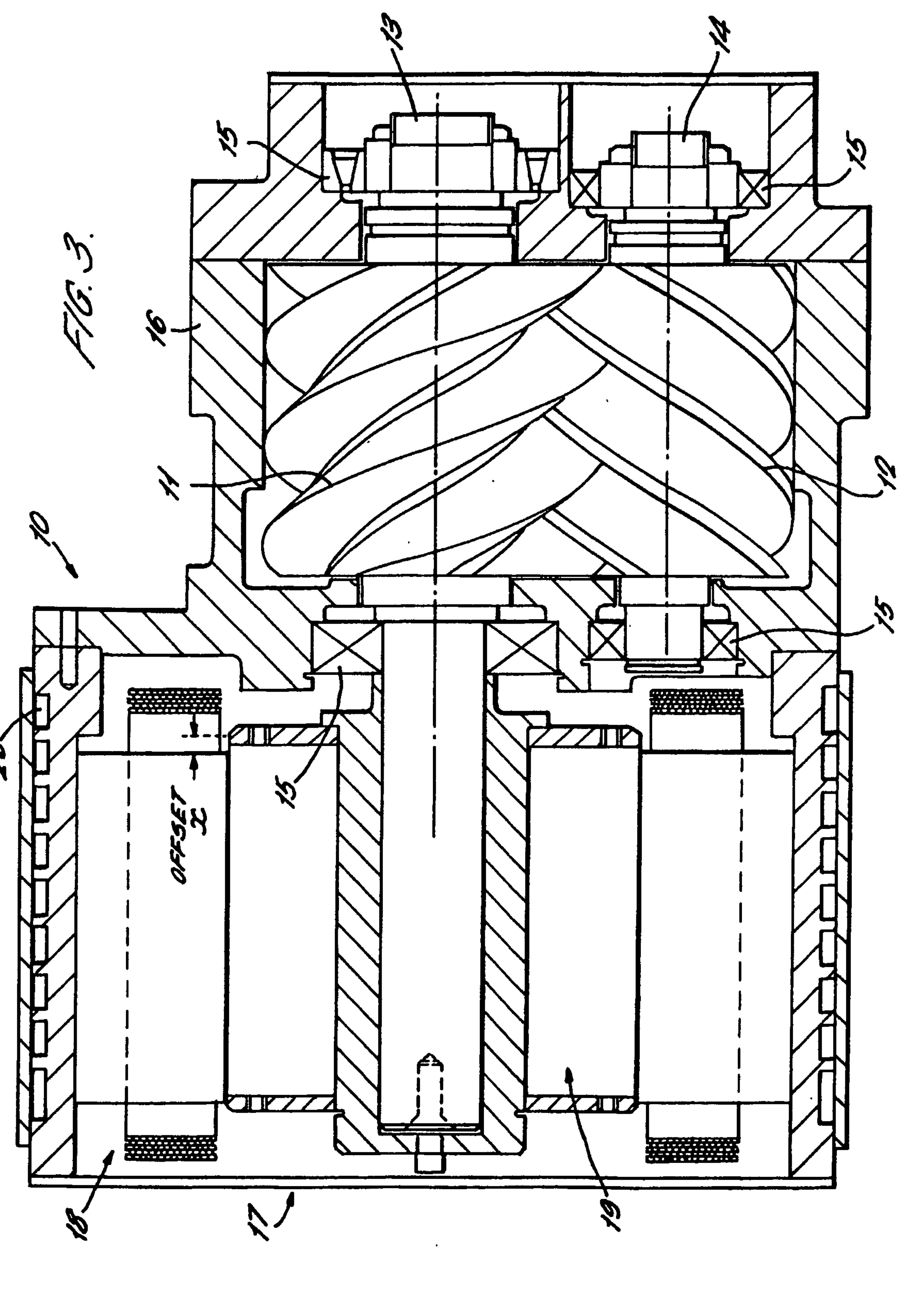


FIG. 3.

INTERNATIONAL SEARCH REPORT

International Application No
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A. CLASSIFICATION OF SUBJECT MATTER
IPC 7 F04C29/00 F01C21/10 H02K7/09 F04C18/16

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
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Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practical, search terms used)

EPO-Internal, PAJ

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[X] Further documents are listed in the continuation of box C.

[X] Patent family members are listed in annex.

° Special categories of cited documents :

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Date of the actual completion of the international search	Date of mailing of the international search report
20 September 2002	27/09/2002

Name and mailing address of the ISA	Authorized officer
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INTERNATIONAL SEARCH REPORT

International Application No
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C.(Continuation) DOCUMENTS CONSIDERED TO BE RELEVANT

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